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MULTIVARIABLE CONTROL OF A MARINE BOILER

Michael Miller

NAVAL POSTGRADUATE SCHOOL
Monterey, California

THESIS

MULTIVARIABLE CONTROL OF A MARINE BOILER

by

Michael Miller

September 1978

Thesis Advisor: T.M. Houlihan

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Multivariable Control of a Marine Boiler

by

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Lieutenant, United States Navy
B.S.E.E., Pennsylvania State University, 1969

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

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An integral output controller is developed for small load changes to a Foster Wheeler ESD-III boiler. The CONSYN program, a coding of modern control algorithms, is utilized to produce a feasible control law for a developed state variable boiler model. The resultant closed loop responses of both a full (10th) order and a reduced (7th) order boiler model are determined using CSMP-III, the IBM simulation language.

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I. INTRODUCTION

As Naval engineers have developed smaller, higher-performance propulsion plants, the requirements for propulsion controls has been transformed from just the design of machinery used for reduced manning to the development of systems needed for safe boiler operations. Likewise, as economic forces emerge more strongly, controls will be required to effect energy conservation. Modern optimal control laws can indeed aid the control engineer in solving both of these problems.

The object of this paper is to develop a linear controller for a marine type boiler using modern optimal control laws. The control problem can be divided into two areas, viz., state estimation and controller design. Only the latter area is investigated here. Extensive use of the programs CONSYN - a coding of modern control algorithms and CSMP - an IBM development, are used in both the controller design and the boiler simulations presented here.

II. BOILER MODEL

The boiler to be controlled is a Foster Wheeler D-type marine boiler. It is an oil fired, two-drum, natural circulation unit having a rated output of 28,800 lbs/hr at 350 psi gauge, with a 1200 F superheater temperature.

The boiler was studied by Whalley [1] in June 1976 and again by Senanikrom [2] in March 1978. They made the following assumptions or simplifications:

a. Superheater

(1) The inertial effects of the superheated steam are neglected.

(2) The superheater tubes are assumed to be a single capacitance with restriction on the drum side and another restriction on the load side.

(3) Desuperheaters are not considered.

b. Downcomer riser loop

(1) Only natural circulation exists.

(2) No boiling takes place in the downcomers.

(3) Vapor and liquid velocities in the riser are identical.

(4) Heat transfer rates to the boiling liquid from the tube walls are proportional to the cube of the temperature difference between the wall and the liquid.

(5) Steam quality is uniform in the riser.

(6) Liquid temperature is always the same as the saturation temperature corresponding to drum pressure.

(7) Downcomer liquid temperature is the same as the drum liquid temperature.

c. Drum

(1) There is no temperature gradient across the drum vapor phase, and the temperature is always the saturation temperature corresponding to the drum pressure.

(2) The liquid phase has no temperature gradient other than across a very thin boundary layer at the drum surface.

(3) Evaporation or condensation rate in the drum is proportional to the difference between liquid and saturation temperatures.

(4) Feedwater temperature is assumed to be constant.

(5) Liquid-level changes due to bubble formation in the drum are neglected.

d. Gas path

(1) The air-fuel ratio is assumed to be constant.

(2) The temperature of combustion gas entering superheater is proportional to the firing rate.

(3) Waterwalls are lumped with the riser-banks.

(4) The heat transfer rate at each tube bank is determined by the tube wall temperature and the average gas temperature.

(5) Inertia of the hot gases is neglected.

(6) Delays due to the heat capacitance of the hot gases are neglected.

(7) All heat transfer is due to turbulent convection and radiation.

Using the laws of Conservation of Mass, Energy and Momentum, 24 non-linear differential equations were developed to describe the boiler operating characteristics. These equations were then linearized about the 50% operating point and arranged in state variable format.

The linearized state variable matrices are listed in Appendix A. The state variable form has ten states, four inputs (throttle valve opening, fuel flow rate, air flow rate, feed flow rate), and four outputs (steam flow from superheater, superheater outlet pressure, steam flow from drum to superheater, drum level).

For simulation purposes the highest order model is desirable. For controller design, however, the lowest order model possible which still closely describes the major characteristics of the boiler is desirable. Since the eigenvalues of the boiler model are widely dispersed (ranging from -85 to -.02 and 0.0) a modal reduction method could have been employed to reduce the order of the model. However, this mathematical method was not employed in the study. Since the boiler controls are designed to control boiler drum pressure (not superheater pressure) and drum level, it was assumed and

verified that deleting the superheater tube wall temperature and the superheater steam temperature as states caused little degradation in the model response. Similarly, since the drum pressure would not be allowed to deviate greatly from the steady state value, the drum and downcomer liquid temperature would remain almost constant. Hence, this state was also eliminated from the model. The final model consisted of seven states (superheater density, steam quality in riser, riser mass-flow rate, downcomer mass-flow rate, riser tube wall temp, drum pressure, drum level). A listing of the reduced order model matrices appears in Appendix B.

III. CONTROL SYSTEM DESIGN

A. OPTIMAL CONTROL

In order to best describe the process of designing the control system a brief resume of 'Optimal Linear Control Theory' is given. The foundation of the theory resides in the works of Kalman and Luenberger on observability, controllability and stability. Controllability and observability are defined as follows: [3]

1. If there is a finite time $t_1 \geq t_o$ and a control $u(t)$, $t \in [t_o, t_1]$, which transfers the state x_o to the origin at time t_1, the state x_o is said to be controllable at time t_o. If all values of x_o are controllable for all t_o, the system is completely controllable, or simply controllable.
2. If by observing the output $y(t)$ during the finite time interval $[t_o, t_1]$ the state $x(t_o) = x_o$ can be determined, the state x_o is said to be observable at time t_o. If all states x_o are observable for every t_o, the system is called completely observable, or simply observable.

The above definitions refer to a system defined in state variable form as

$$\dot{\underset{\sim}{x}} = \underset{\sim\sim}{A}\underset{\sim}{x} + \underset{\sim\sim}{B}\underset{\sim}{u}$$

$$\underset{\sim}{y} = \underset{\sim\sim}{C}\underset{\sim}{x} + \underset{\sim\sim}{D}\underset{\sim}{u}$$

where $\underset{\sim}{x}$ is a vector of the states of the system;
$\underset{\sim}{u}$ is a vector of the inputs of the system and;
$\underset{\sim}{y}$ is a vector of the outputs of the system.

One final definition is required, that of the performance measure. This is the criteria by which the "goodness" of a control design is measured. It is usually of the form $J = f(\underset{\sim}{x},\underset{\sim}{u},t)$.

The optimal control problem reduces to finding a control, $\underset{\sim}{u}$, which causes the system $\dot{\underset{\sim}{x}} = \underset{\sim\sim}{A}\underset{\sim}{x} + \underset{\sim\sim}{B}\underset{\sim}{u}$ to follow a course which minimizes a performance measure J. In linear systems, if the system is observable and controllable, then there exists a $\underset{\sim}{u}$ such that $\underset{\sim}{u} = -\underset{\sim\sim}{G}\underset{\sim}{x}$ which minimizes the performance measure, J.

The most common performance measures are the following:

$$J_1 = \int \underset{\sim}{u}^T \underset{\sim}{R}\ \underset{\sim}{u}\ dt \quad \text{(minimum inputs)}$$

$$J_2 = \int \underset{\sim}{x}^T \underset{\sim}{Q}\ \underset{\sim}{x}\ dt \quad \text{(minimum excursions)}$$

$$J_3 = \int (\underset{\sim}{x}^T \underset{\sim}{Q}\ \underset{\sim}{x} + \underset{\sim}{u}^T \underset{\sim}{R}\ \underset{\sim}{u})\ dt \quad \text{(combined minimums)}$$

The performance index used in this design is J_3. If the control is unconstrained and if the weighting matrices $\underset{\sim}{Q}$ and $\underset{\sim}{R}$ are positive semi-definite and positive definite respectively,

then there are numerical methods for calculating the $\underset{\sim}{G}$ matrix. The only decision facing the design engineer is the relative importance of the states and the controls (i.e., the values of $\underset{\sim}{Q}$ and $\underset{\sim}{R}$). This is no easy task, for as the size of the system increases, so do the design variables and the permutations of solutions. Several general guidelines are suggested for initial designs. One of the best is the (1/delta square) normalization. In this method the values of the $\underset{\sim}{Q}$ and $\underset{\sim}{R}$ matrices are:

$$\underset{\sim}{Q} = \begin{bmatrix} \frac{1}{(\Delta X_1)^2} & 0 & \cdots & 0 \\ 0 & \frac{1}{(\Delta X_2)^2} & \cdots & \vdots \\ \vdots & \vdots & & \vdots \\ 0 & \cdots & \cdots & \frac{1}{(\Delta X_n)^2} \end{bmatrix} \qquad \underset{\sim}{R} = \begin{bmatrix} \frac{1}{(\Delta U_1)^2} & 0 & \cdots & 0 \\ 0 & \frac{1}{(\Delta U_2)^2} & \cdots & \vdots \\ \vdots & \vdots & & \vdots \\ 0 & \cdots & \cdots & \frac{1}{(\Delta U_m)^2} \end{bmatrix}$$

where "Δ" is the maximum expected deviation from a given operating point.

Since most systems have a limited amount of controls (i.e., maximum fuel flow, maximum valve opening), the control law, $\underset{\sim}{u} = -\underset{\sim}{\underset{\sim}{G}}\underset{\sim}{x}$ is only valid if $\underset{\sim}{u}$ is within the constraints of the system. If this is not the case the designer has two options. He can make a non-linear controller such that $\underset{\sim}{u} = -\underset{\sim}{\underset{\sim}{G}}\underset{\sim}{x}$

for all $\underset{\sim}{u}$ within the constraint boundary and $\underset{\sim}{u}$ equals the constraints for all other cases, or he can choose different weighting matrices such that the controls and states meet all constraints. The last method was employed in this study.

B. CONSYN

The CONSYN program developed by Lt. M. Dundics [4] was designed to aid the engineer in the iterative process of designing a controller. Inputs are the state variable model, initials guesses for $\underset{\sim}{Q}$ and $\underset{\sim}{R}$ and the system constraints. The program then varies the $\underset{\sim}{Q}$ and $\underset{\sim}{R}$ matrices to obtain a minimum J which meets all of the system constraints. If the initial $\underset{\sim}{Q}$ and $\underset{\sim}{R}$ matrices do not produce a feasible design the program will change the matrices so as to satisfy all constraints. The outputs from the program are two sets of gains, $\underset{\sim}{L}$ and $\underset{\sim}{H}$. $\underset{\sim}{L}$ is the matrix of state regulator feedback gains and $\underset{\sim}{H}$ is the matrix of integral feedback gains.

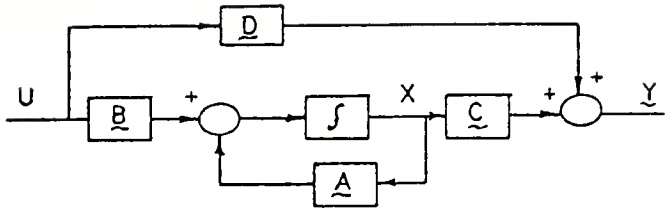
Both Michael [5] and Tysso [6] recommend integral (reset) control. Tysso points out that integral control affords a soft or "bumpless" transfer from a conventional backup system to the multivariable mode. Michael states that integral control is less sensitive to degradations of the system. This insensitivity property reduces the differences between full order and reduced order model responses.

To obtain integral control, the original model with n inputs (Fig. 1a) is augmented by n integrators. This augmented system (Fig. 1b) can be rearranged into the form shown

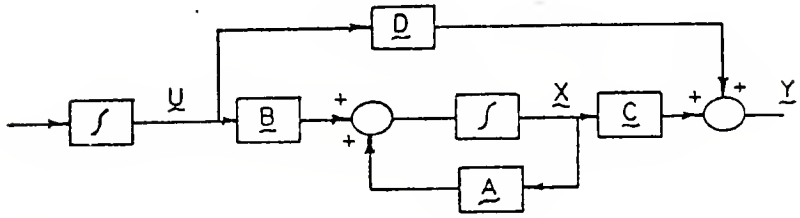
in Fig. 1c. The optimal control law reduces to $\underset{\sim}{u}^* = -\underset{\sim}{G}^*\underset{\sim}{x}^*$. By ordering the outputs such that the first n outputs are the quantities to be regulated, a simple analysis will transform the optimal state regulator solution into the optimal integral control system as shown in Fig. 2. $\underset{\sim}{R}$ in this case is the set point or reference vector.

In using the CONSYN routine two minor problems were encountered. In order to check for violations of constraints, the system must be simulated for a duration of time greater than the desired settling time. CONSYN used a discrete solution, $\underset{\sim}{x}(kT + T) = \exp \underset{\sim}{F}T\ \underset{\sim}{x}(kT)$, where T is a fixed step size and $\underset{\sim}{F} = (\underset{\sim}{A}-\underset{\sim}{B}\underset{\sim}{G})$ is the closed loop gain of the system. This simulation is very efficient for most systems. The boiler model however, is one exception since it is a "stiff" system. It requires a very small time increment to produce the initial response, but could incorporate a much larger step size to produce the mid-phase and terminal phase of the system response. The use of a fixed step integration routine caused an inefficient use of computer time. For this study this fixed step integration routine was replaced by a variable step routine, Dvoger, an IBM-IMSL routine.

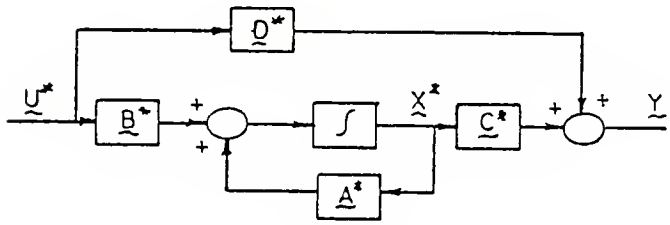
Moreover, the CONSYN routine used a Kleinman technique to derive the regulator gains, $\underset{\sim}{G}$. DiPietro [7] noted that this method exhibits numerical problems if a zero eigenvalue exists. The model has one zero eigenvalue (associated with drum level) and numerical overflows and underflows were experienced. For



1a



1b



$$\underset{\sim}{X}^* = \begin{bmatrix} \underset{\sim}{X} \\ \hline \underset{\sim}{U} \end{bmatrix} \qquad \underset{\sim}{A}^* = \left[\begin{array}{c|c} \underset{\sim}{A} & \underset{\sim}{B} \\ \hline \underset{\sim}{O} & \underset{\sim}{O} \end{array}\right]$$

$$\underset{\sim}{B}^* = \begin{bmatrix} \underset{\sim}{O} \\ \hline \underset{\sim}{I} \end{bmatrix} \qquad \underset{\sim}{C}^* = \left[\begin{array}{c|c} \underset{\sim}{C} & \underset{\sim}{D} \end{array}\right]$$

$$\underset{\sim}{D}^* = \begin{bmatrix} \underset{\sim}{O} \end{bmatrix}$$

1c

Figure 1 – Augmented Optimal Control System Development

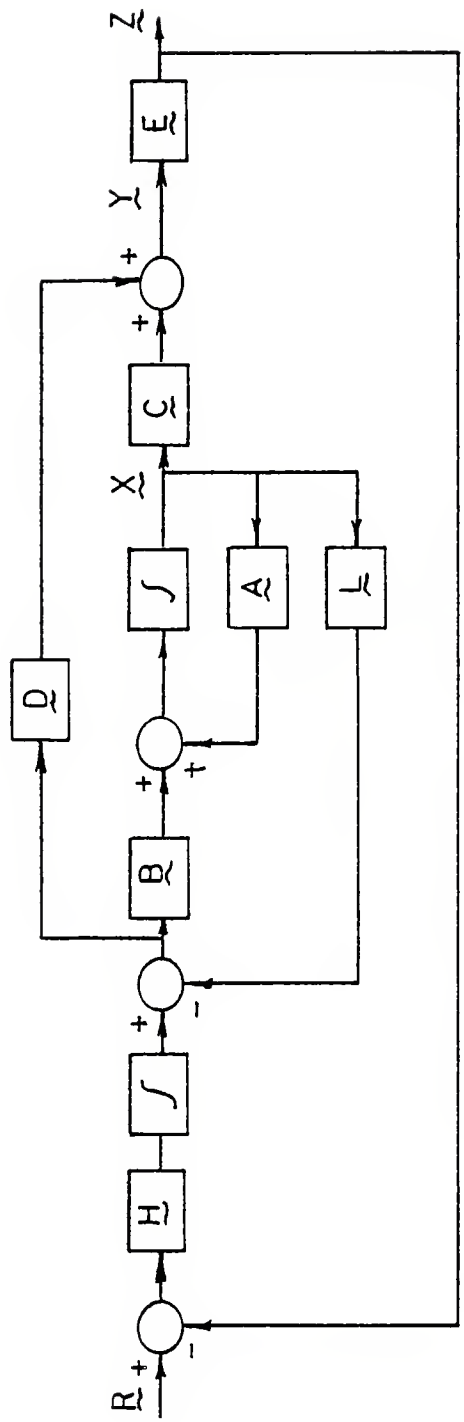


Figure 2 — Optimal Integral Control System

this study the Kleinman method was replaced by an eigenvalue solution of the matrix Ricatti equation.

C. SYSTEM CONSTRAINTS

Since the model is of a particular boiler and not of a complete system the following assumptions were made concerning the auxiliary support system:

1. A constant fuel to air ratio was maintained — stoichimetric plus 15% excess air.

2. Time delays in sensors and actuators were ignored.

3. Since system response would be limited by response of air flow, it was assumed that the maximum rate of change was one percent of actual flow rate.

4. Maximum allowed pressure deviation was five lbs/in^2.

5. Maximum allowed water level deviation was one inch.

All control tests perturbed the throttle valve such as to cause a steam flow change of 5% (from 50% to 55%). The valve is scheduled to make this change in 10 seconds (starting at a time of one second) with no change in feed flow or fuel and air flow.

Figs. 3 and 4 show the open loop responses of the reduced order model to the throttle control perturbation. Figs. 5 and 6 show the open loop responses of the full order model to the throttle control perturbation. As these responses indicate, the system reaches a new operating point with a lower drum pressure and a lower drum level. It's to be noted that the differences betwen the open loop responses of the full and

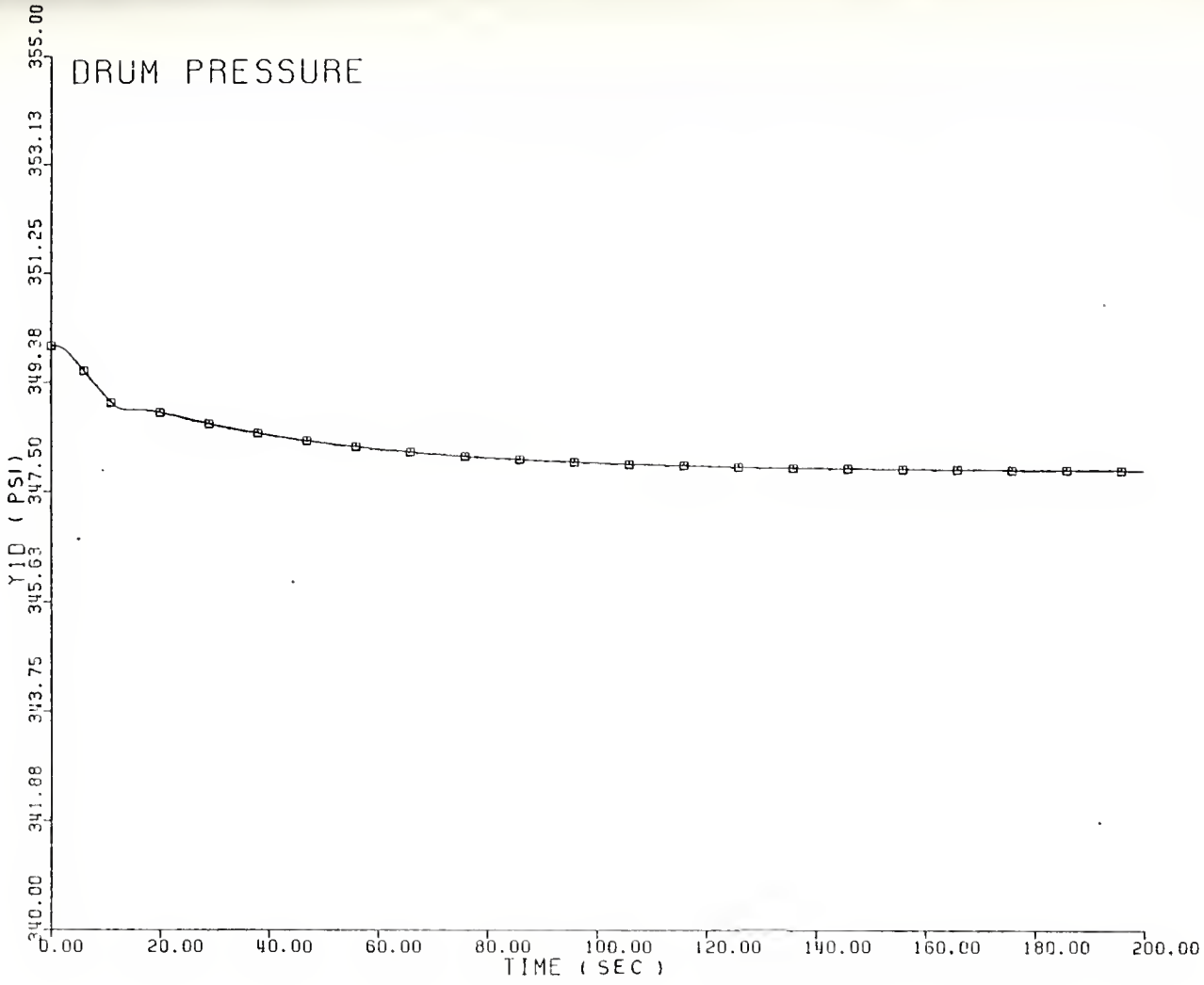


Figure 3 – Drum Pressure Variation (Open Loop, Reduced Order Model)

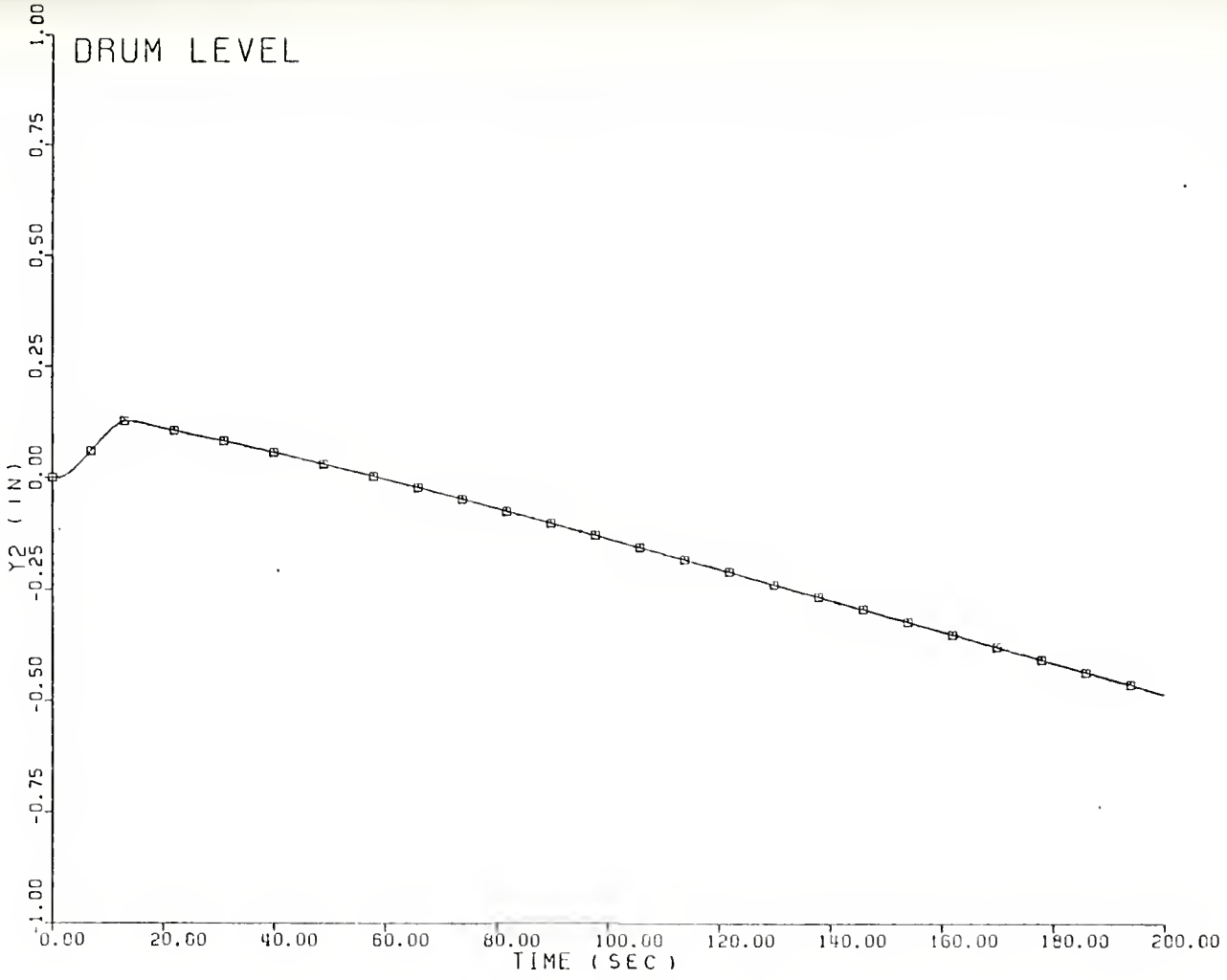


Figure 4 – Drum Level Change (Open Loop, Reduced Order Model)

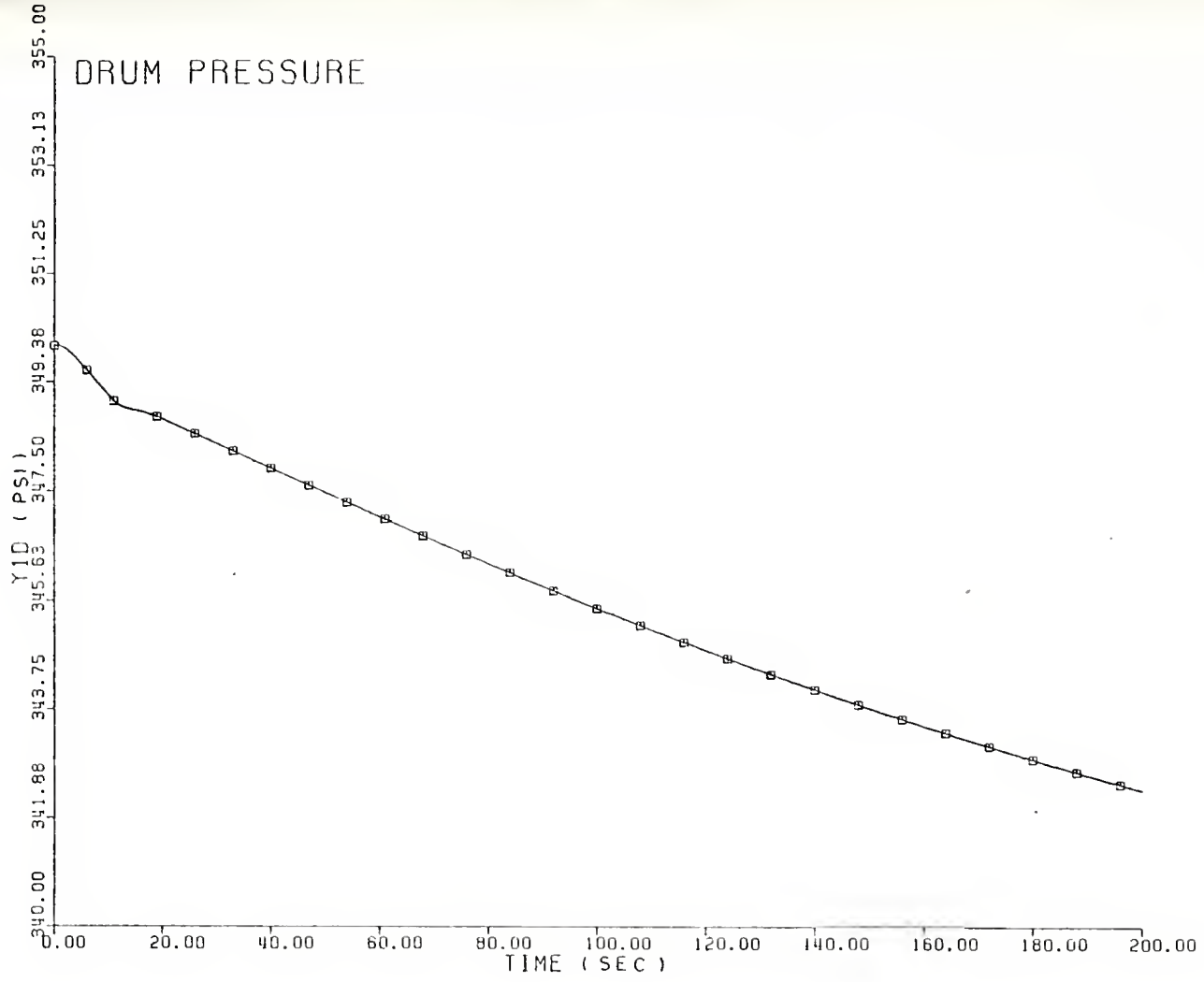


Figure 5 – Drum Pressure Variation (Open Loop, Full Order Model)

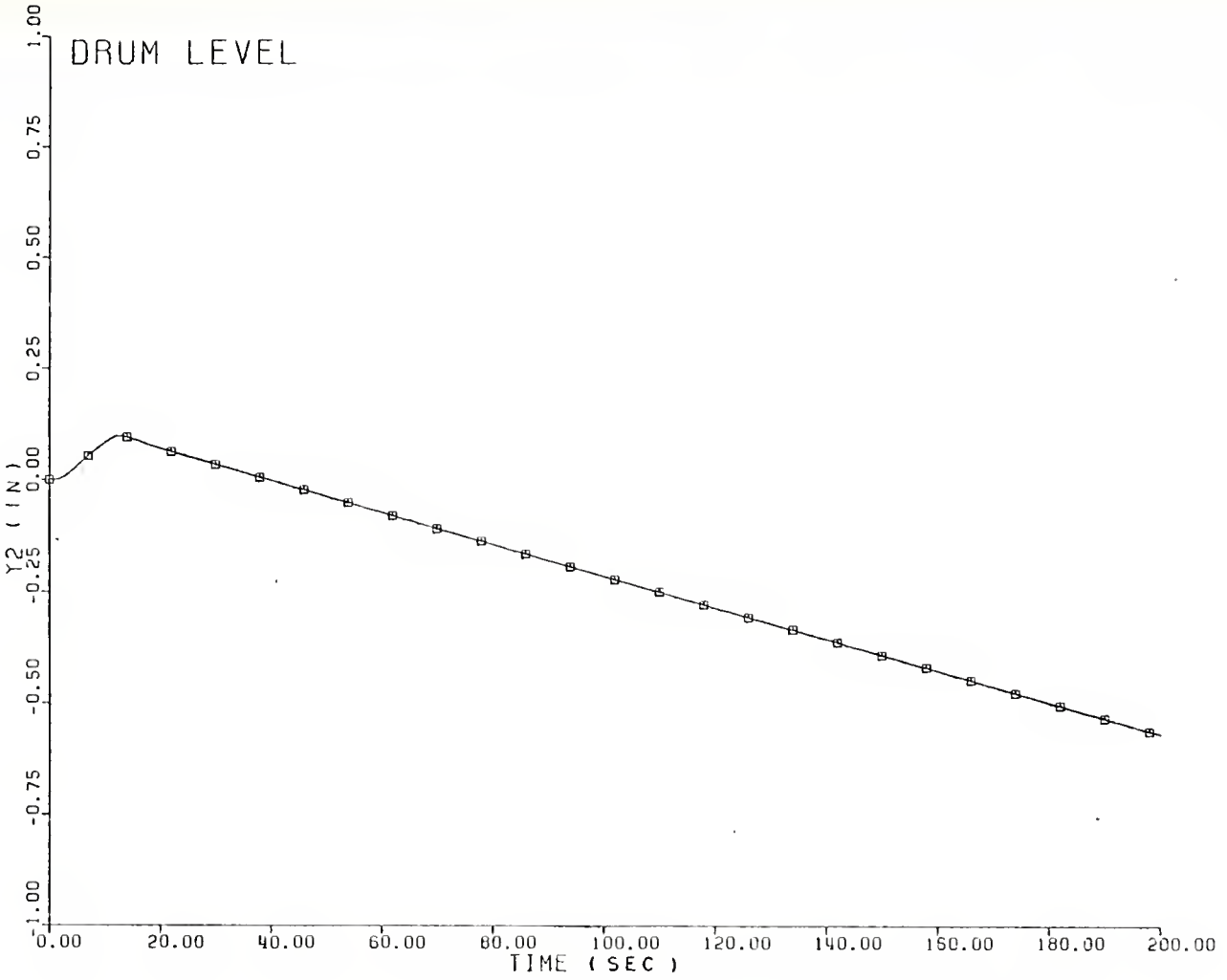


Figure 6 — Drum Level Change (Open Loop, Full Order Model)

reduced order models are enhanced by the scaling presented in the CSMP graphical outputs. In realistic terms, these differences are very minor.

The following values were used to develop an initial weighing matrix:

Δ pressure = 720 lbf/ft^2

Δ level = 0.5 ft

Δ fuel rate = 0.01 lbm/sec

Δ feed rate = 0.7 lbm/sec

Δ fuel rate change = 0.0032 lbm/sec/sec

Δ feed rate change = 0.1 lbm/sec/sec

The control system developed using these initial weighing matrix values was applied to both the reduced order model and the full order model of the boiler. Figures 7-10 show the closed loop response characteristics of the reduced order model. Figures 11-14 show the closed loop response characteristics of the full order model. For both models, the drum pressure returns to normal within 60 seconds with only a one psi loss in pressure during the transient. The drum water level remained essentially at the normal steaming level during the entire evolution.

The additional transient observed to occur (after 60 seconds) when the full order model air flow rate (Fig. 14) is compared to the reduced order model air flow rate (Fig. 10) is due to the additional energy transferred by the temperature

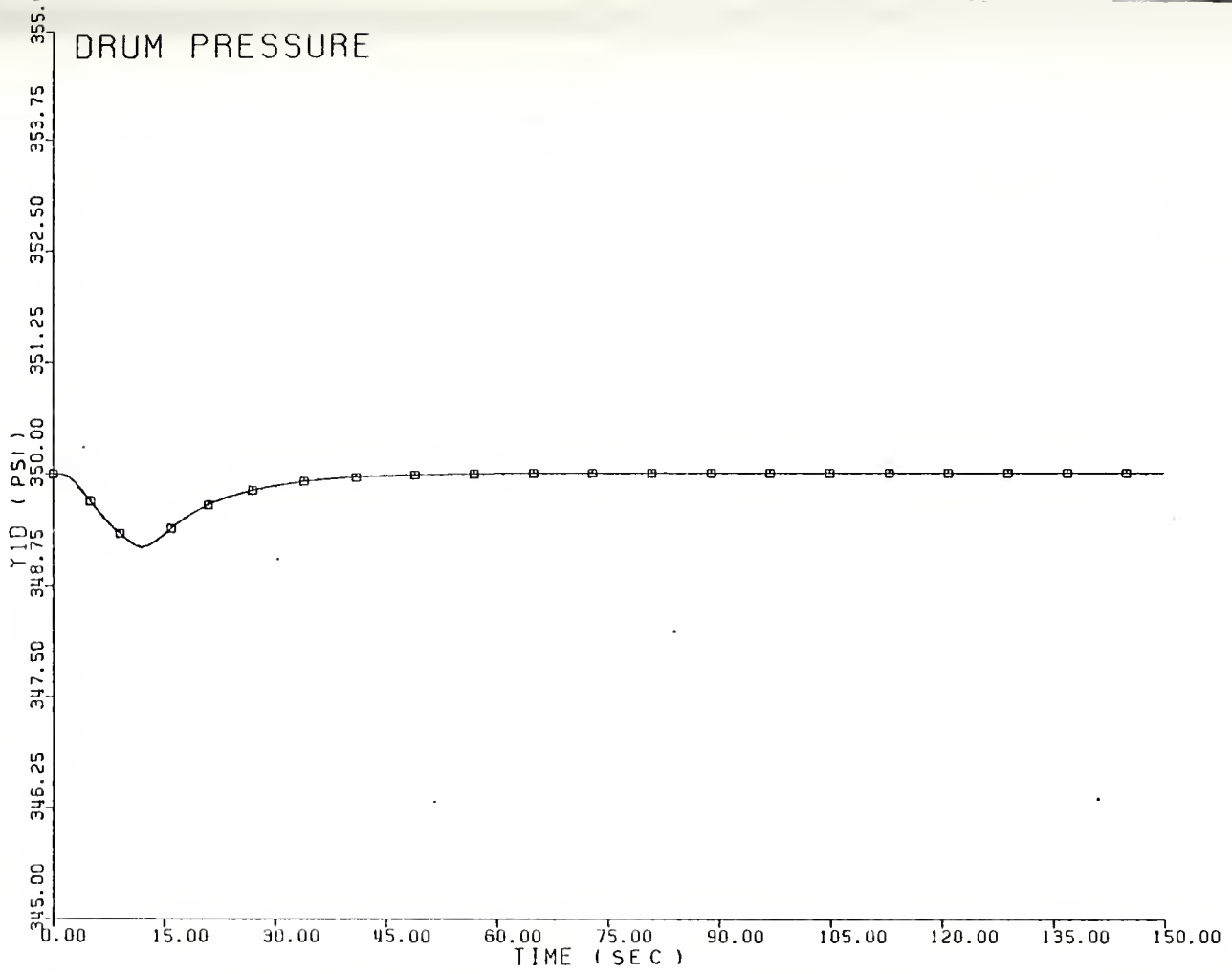


Figure 7 — Drum Pressure Variation (Closed Loop, Reduced Order Model, J = 1.13)

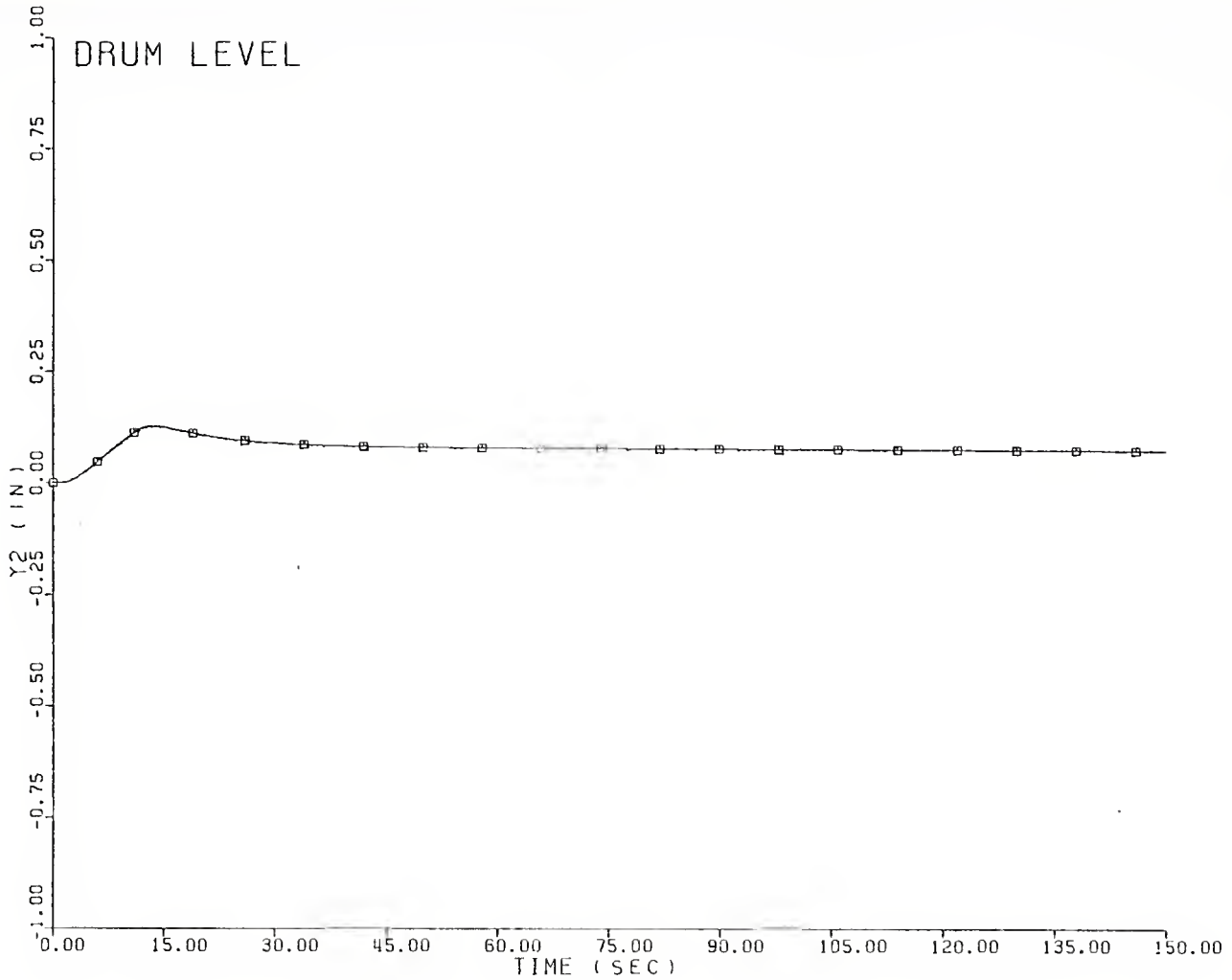


Figure 8 — Drum Level Change (Closed Loop, Reduced Order Model, J = 1.13)

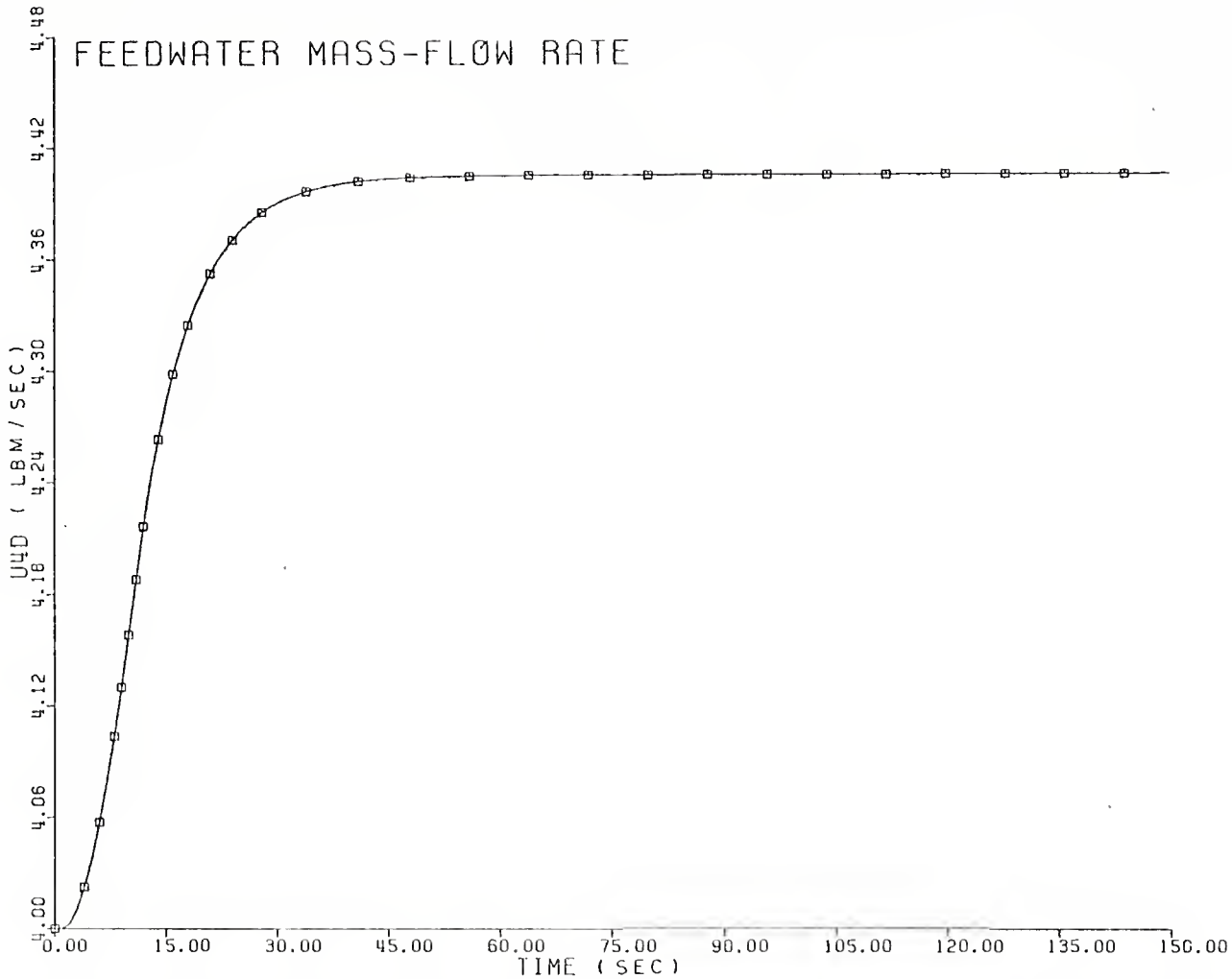


Figure 9 — Feed Flow Rate (Closed Loop, Reduced Order Model, J = 1.13)

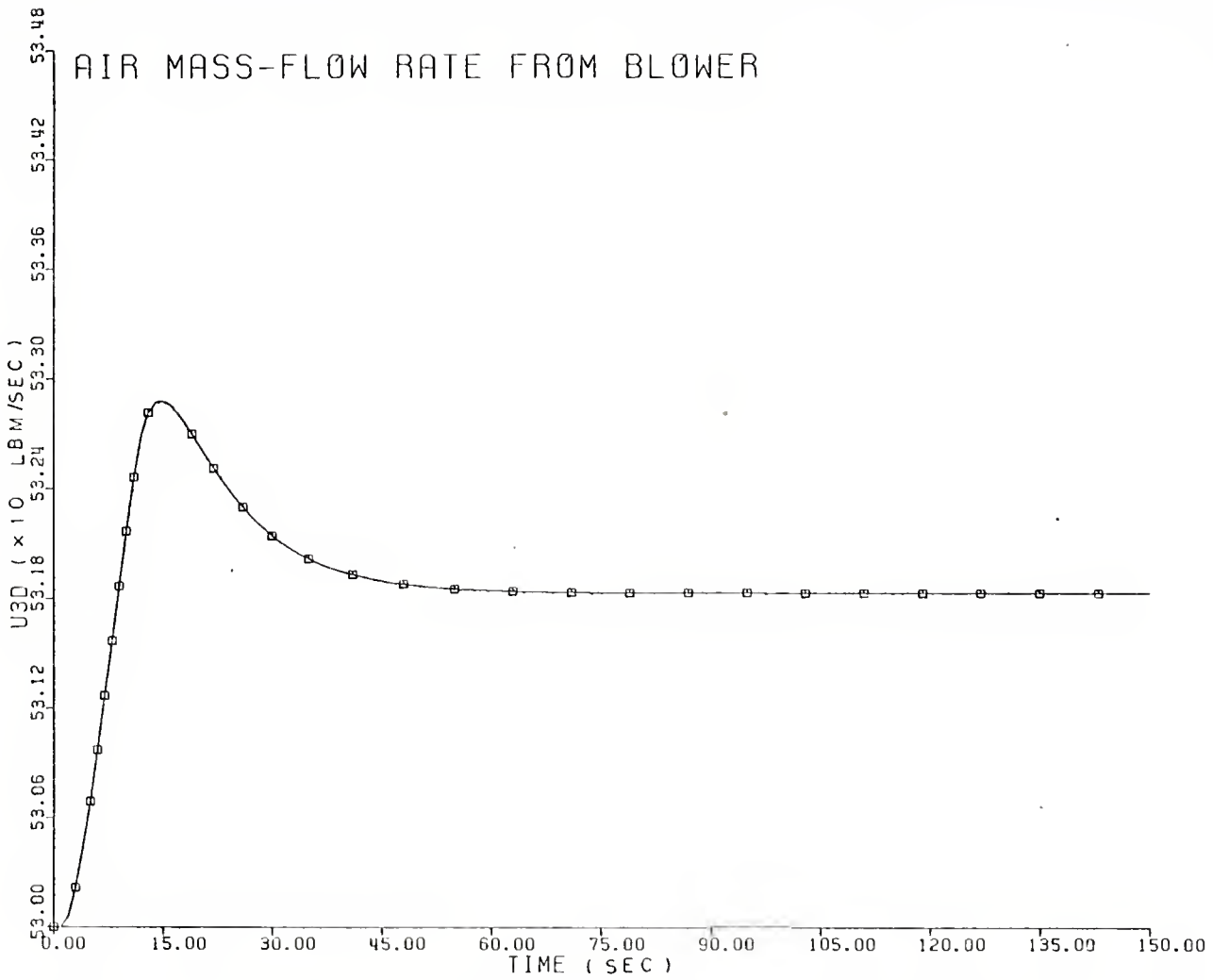


Figure 10 – Air Flow Rate (Closed Loop, Reduced Order Model, J = 1.13)

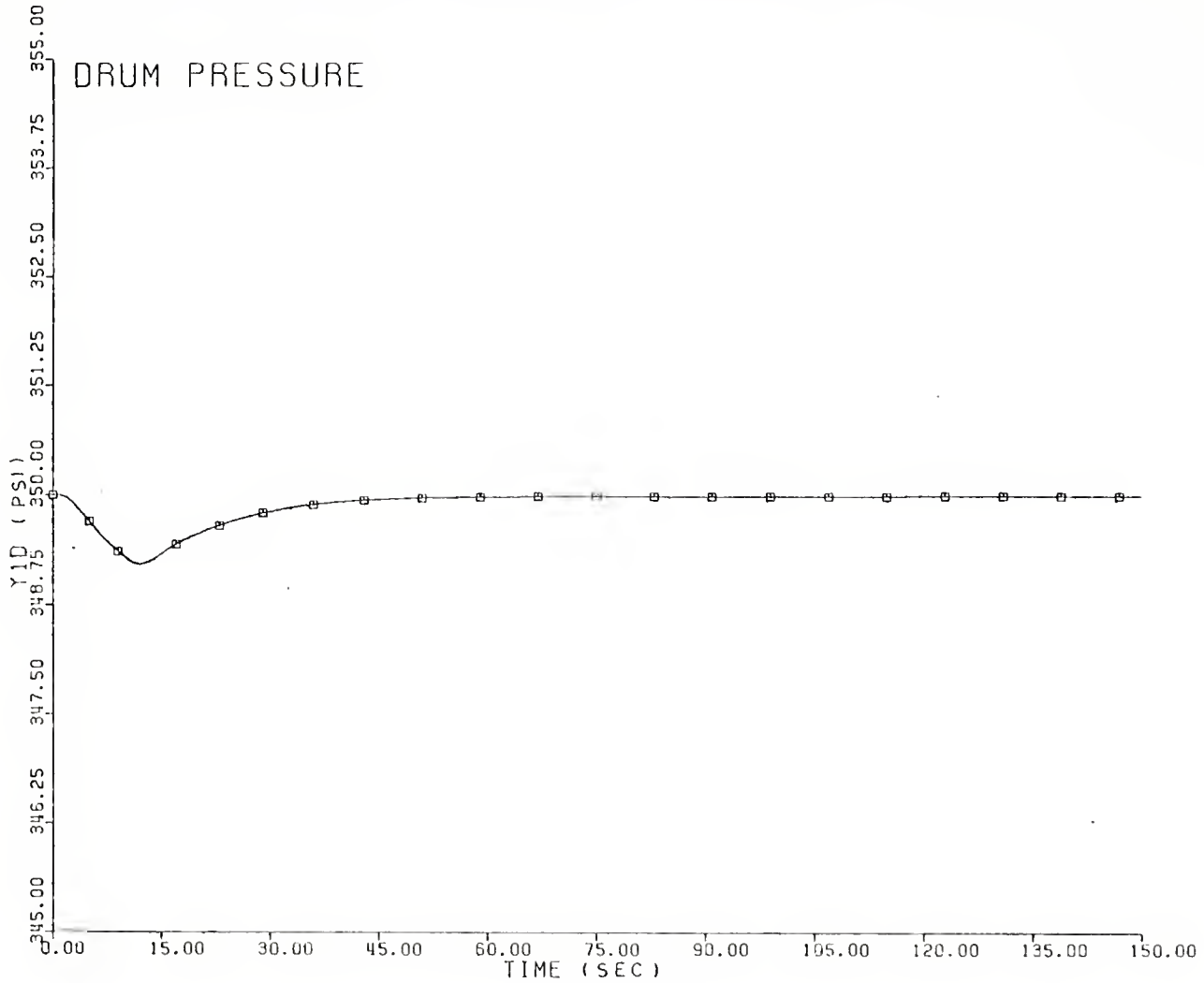


Figure 11 — Drum Pressure Variation (Closed Loop, Full Order Model, J = 1.13)

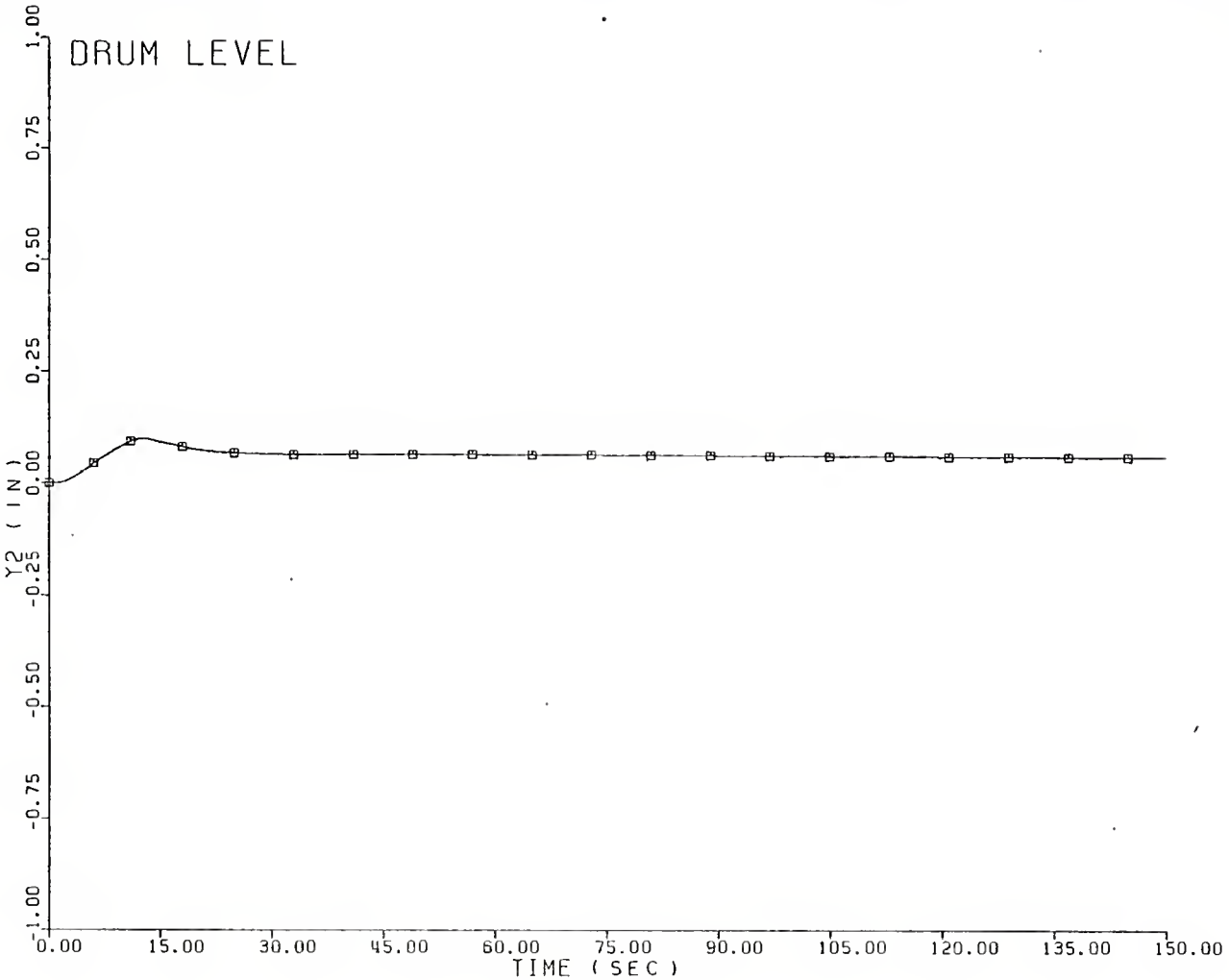


Figure 12 — Drum Level Change (Closed Loop, Full Order Model, J = 1.13)

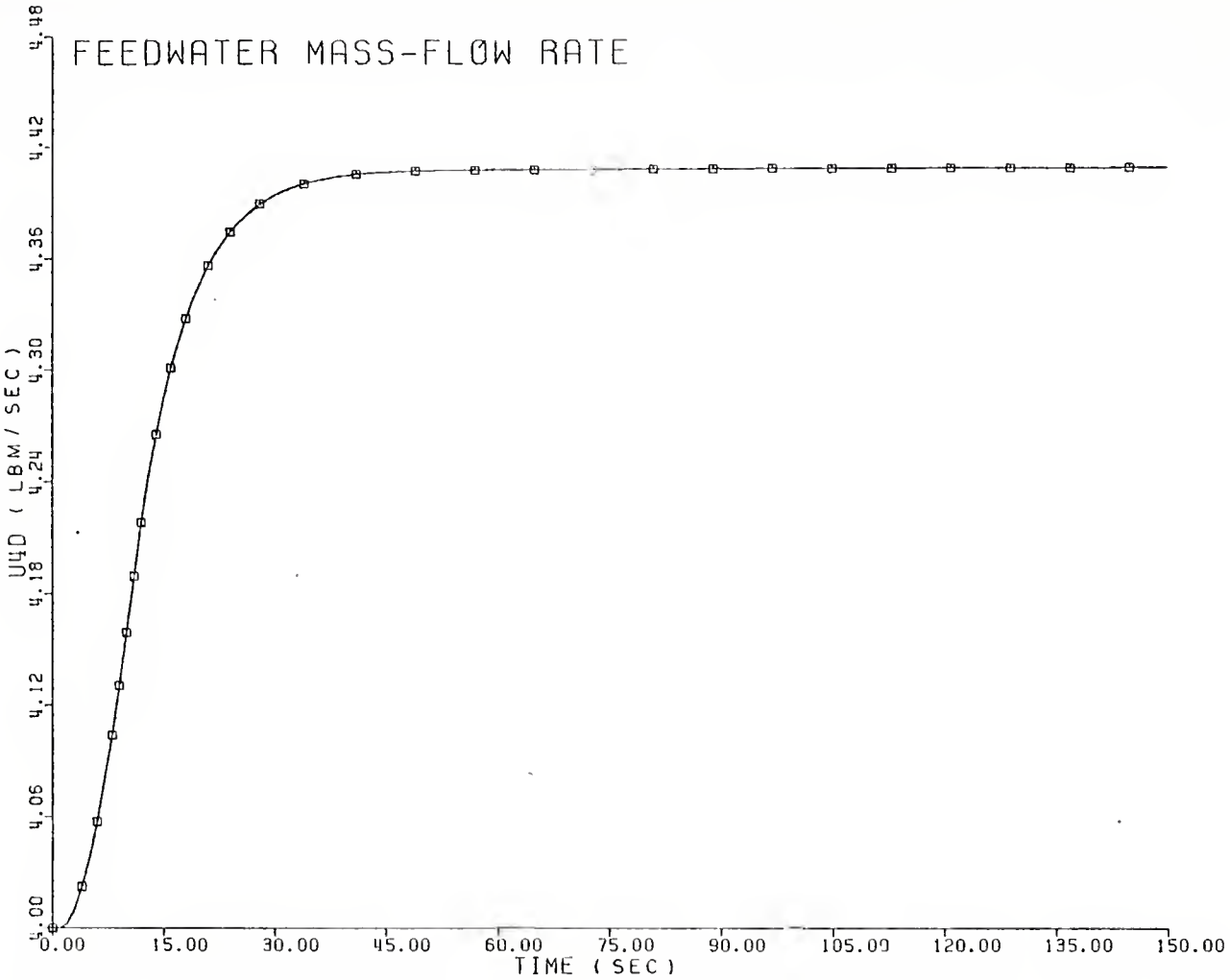


Figure 13 – Feed Flow Rate (Closed Loop, Full Order Model, J = 1.13)

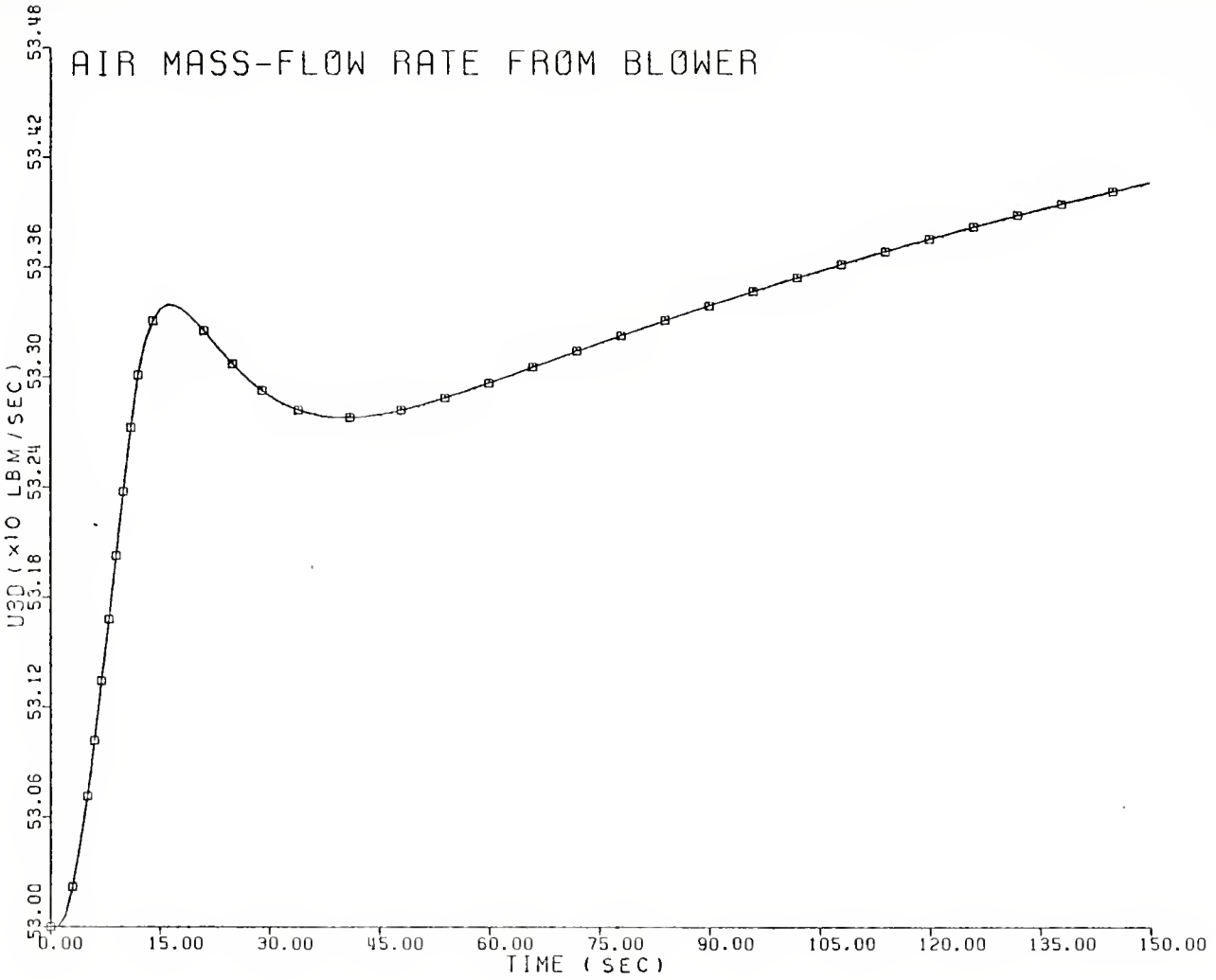


Figure 14 — Air Flow Rate (Closed Loop, Full Order Model, J = 1.13)

differences which appear in the full order model and are neglected in the reduced order model. Since the reduced order model produces drum level responses which are identical to the full order model responses, an additional transient is not observed when comparing the feed flow rate for the full order model (Fig. 13) to that of the reduced order model (Fig. 9). In this latter instance, mass conservation considerations rather than energy conservation considerations apply.

The CONSYN program modified the initial weighing matrix and reduced the performance index value from 1.13×10^6 to 4.7×10^5 subject to the specified constraints on states and inputs for a settling time of 150 seconds. This value for settling time was set by boiler flex test considerations. The outputs and the feed flow rate responses (as shown in Figures 15-18), were essentially the same as before (see Figures 11-14). The only noticeable change was in air flow rate (Figures 14 and 18). The initial peak in air flow rate was not as high (3.25 vs. 3.36) and the relative minimum was higher (2.81 vs. 2.79) for the latter index value. The final response ($J = 4.7 \times 10^5$) being less oscillatory would place less demands on the machinery and thus produces a more reliable system. Again, it is to be noted that the scaling of the CSMP produced response curves enhances the slight variations existent in the operating characteristics. In reality, these differences are very minor in nature.

Further attempts to decrease settling time constraints were limited due to uncertainties in the values of rate

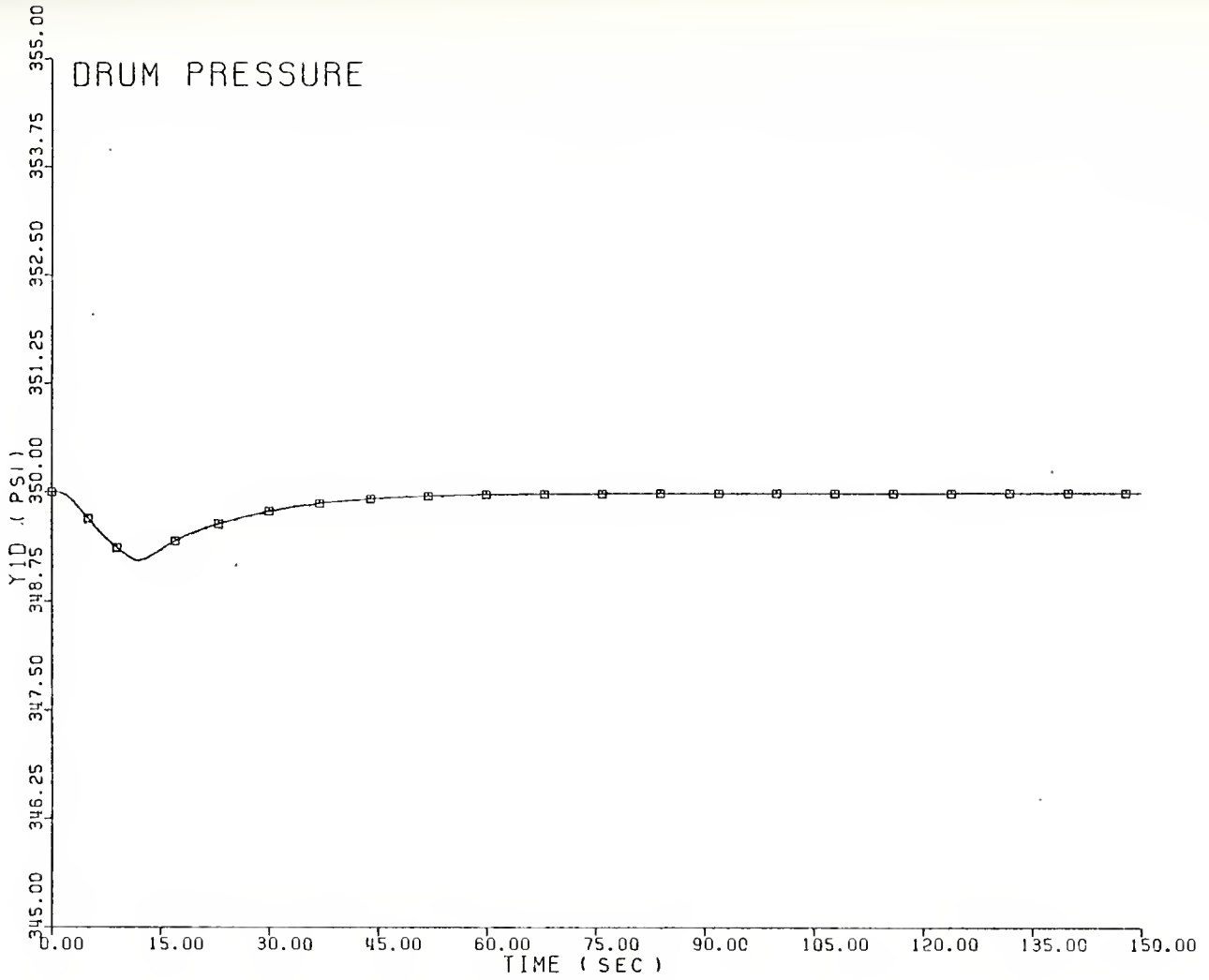


Figure 15 – Drum Pressure Variation (Closed Loop, Full Order Model, J = 0.47)

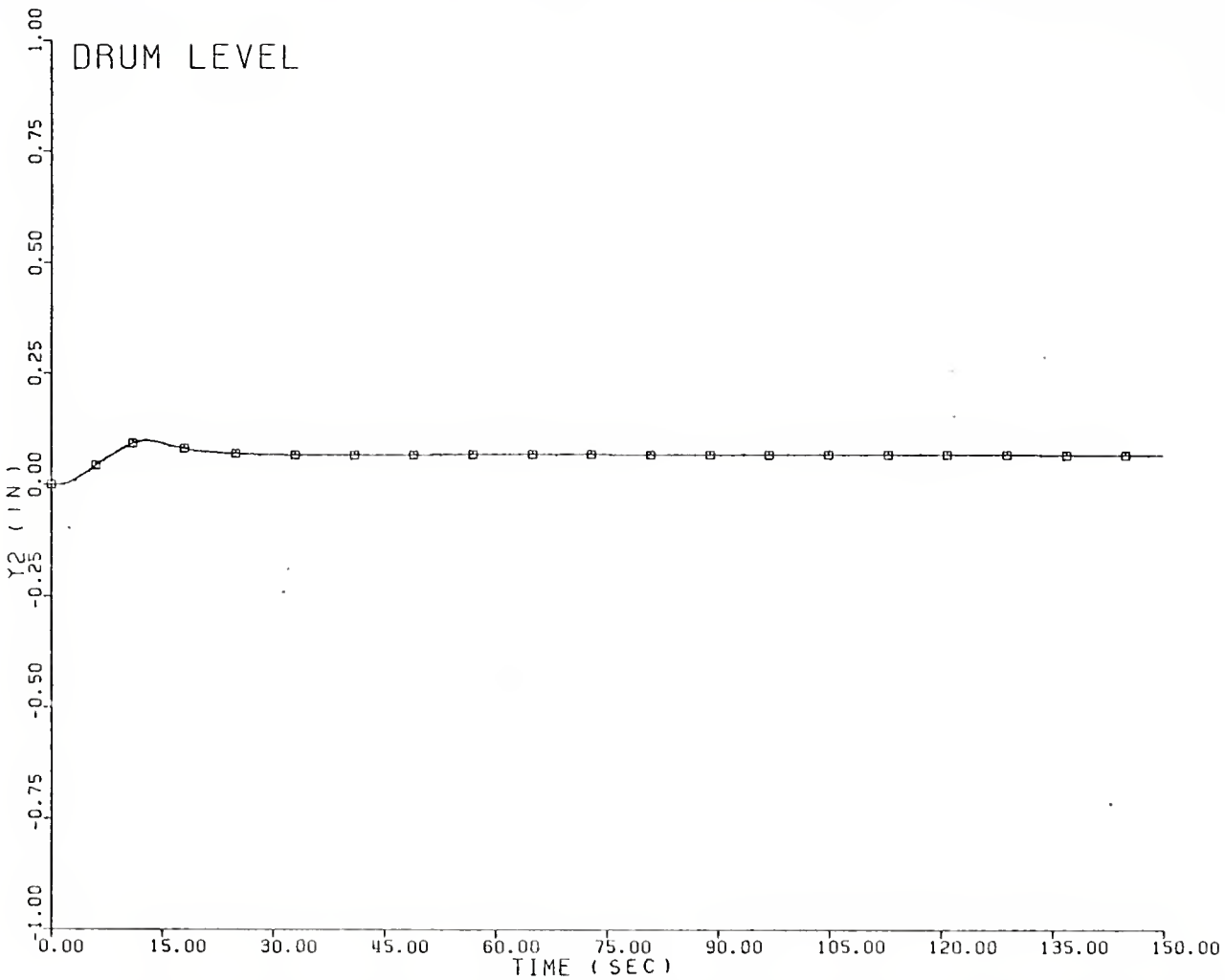


Figure 16 – Drum Level Change (Closed Loop, Full Order Model, J = 0.47)

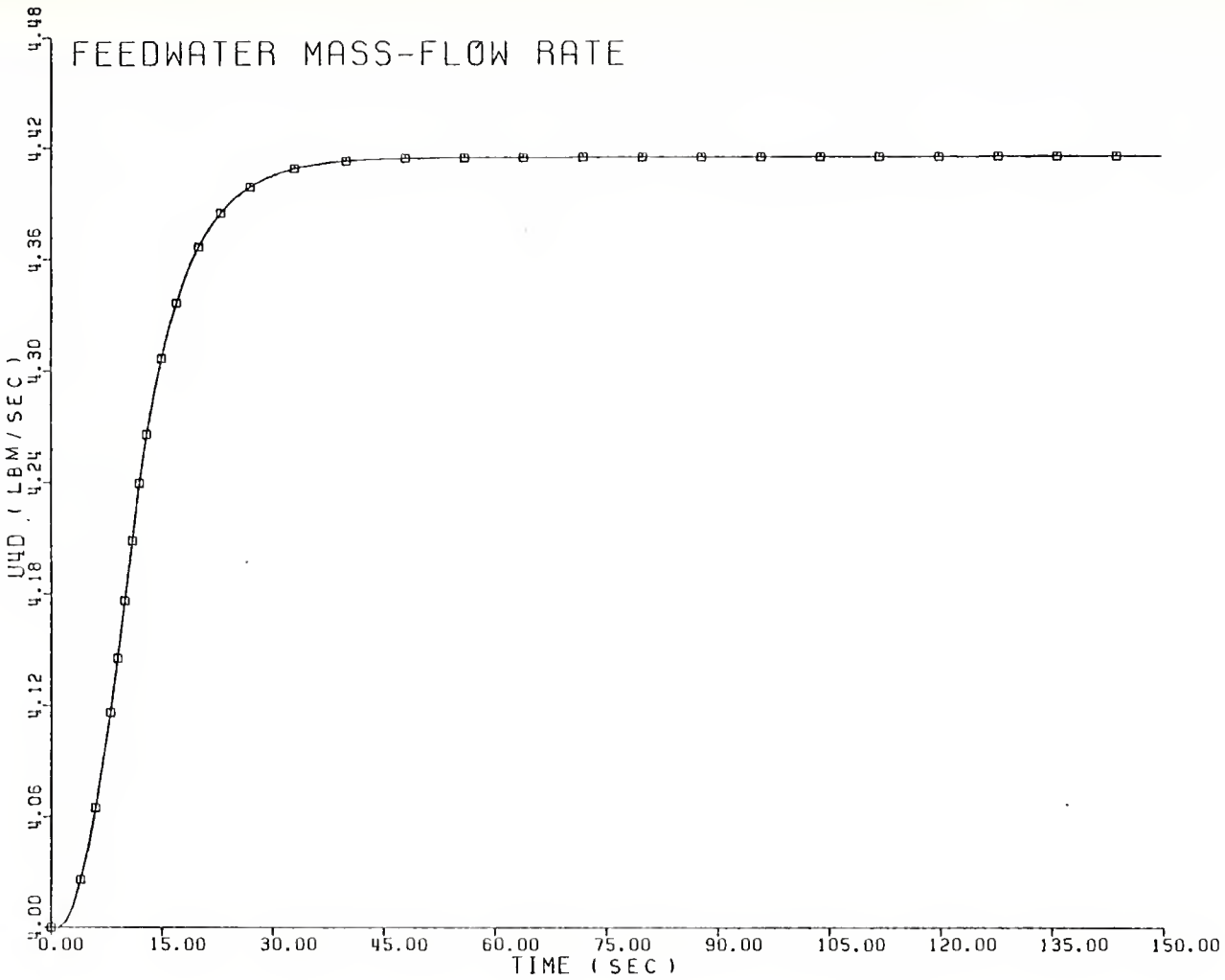


Figure 17 – Feed Flow Rate (Closed Loop, Full Order Model, J = 0.47)

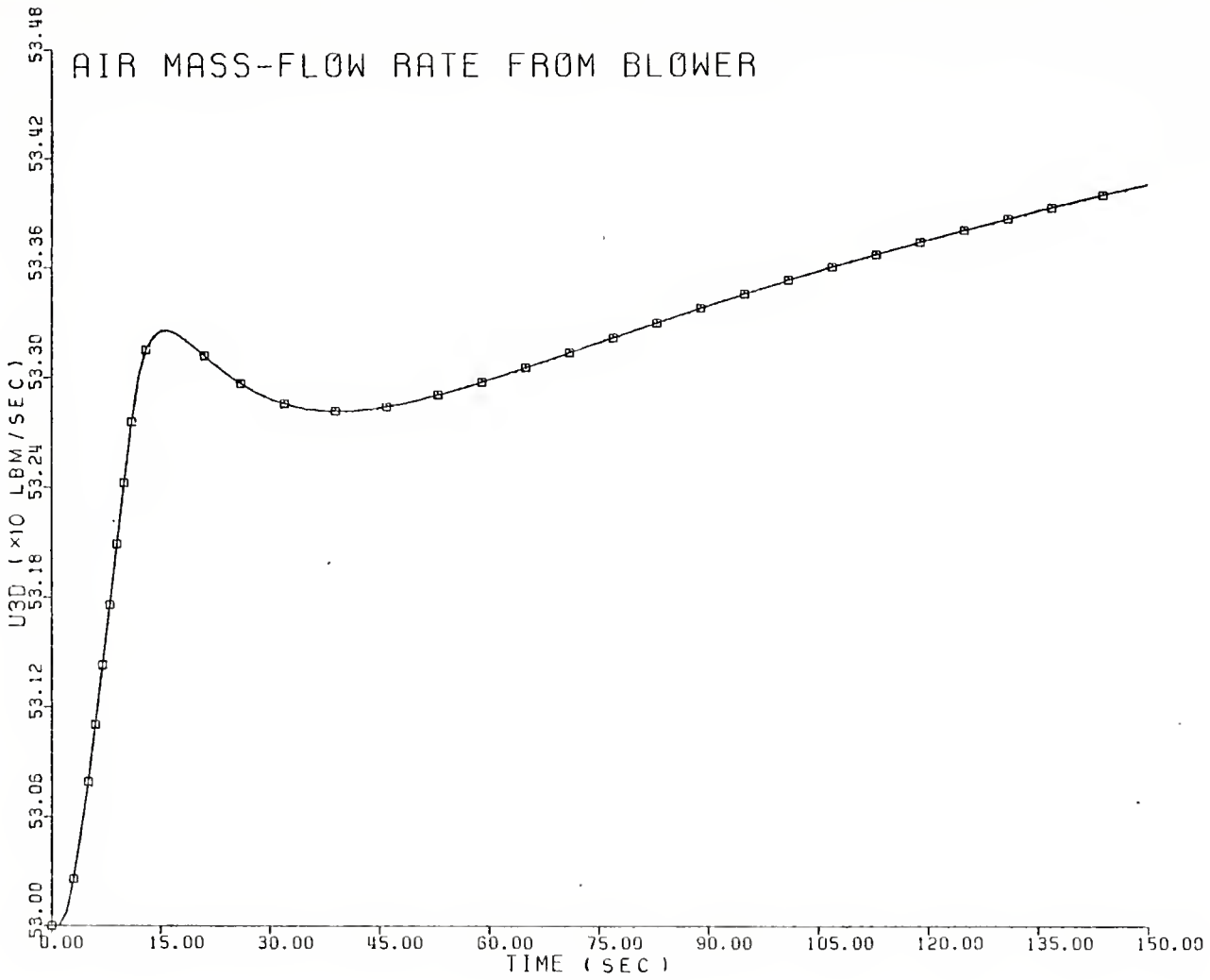


Figure 18 — Air Flow Rate (Closed Loop, Full Order Model, J = 0.47)

changes that would apply to the pumps and blowers of the boiler system under consideration. In fact, limited knowledge of the approximate physical and thermodynamical characteristics of the ESD-III installation continually hampered this study.

IV. CONCLUSIONS AND RECOMMENDATIONS

The CONSYN program is a valuable tool for the control engineer. Moreover, integral control is relatively insensitive to state changes. The only differences noted between controlling the full order model and the reduced order model were in the responses of the fuel/air flow input rates. Large demand changes were not simulated because the model is only valid for small perturbations about the 50% steaming rate. In a typical boiler flex test, the boiler is required to reach steady state in three minutes following a steam demand change from 20% to 85%, in 45 seconds. A settling time of 150 seconds was chosen to be comparable with a boiler flex test. This constraint was never exceeded. A settling time of 60 seconds was actually met since only the air-fuel flow input rate for the full order model exceeded this settling time requirement. However, the rate change here is very small and as such is not deemed detrimental to the system.

In order to do a complete boiler control design the following extensions to this study are required:

1. A non-linear model is required which is valid over the 20% to 100% steaming conditions. Such a model was not constructed for this study because complete operating information on the ESD-III boiler installation could not be readily assembled.

2. The non-linear model must be linearized at various operating points to obtain a group of model characteristics for controller design.

3. If necessary, the order of the linearized model must be reduced to the size necessary for CONSYN (maximum number of states and inputs is ten; maximum number of outputs is ten) or the storage requirements of the program must be increased to meet the system requirements.

4. System observers must be designed to produce values for those states not normally measured.

APPENDIX A

* FULL ORDER MODEL MATRICES ELEMENTS

A. A MATRIX.

A11 = -1.2436020E 01	A12 = -3.1416850E-01	A18 = 1.3159210E-03
A21 = 1.7138750E -3	A22 = -1.6096850E 00	A23 = 5.1956340E 00
A28 = -1.8383510E-02	A31 = 4.4709970E 01	A32 = 1.4683690E-01
A33 = -3.4865910E-02	A38 = -4.7957190E-04	A41 = -3.4339410E 00
A42 = -8.6752210E-03	A44 = -4.7662320E 00	A45 = -1.1941460E-03
A46 = 7.0738900E-04	A47 = -7.2198050E-04	A48 = 2.7080380E-04
A49 = 1.0494110E-00	A51 = -4.9093270E 05	A52 = -1.2402500E 03
A54 = -6.3842140E 05	A55 = -1.5757540E 02	A56 = 9.3258980E 02
A57 = 9.9146950E 01	A58 = 4.0967170E-01	A59 = 1.4977520E 03
A61 = -7.0894060E 04	A62 = -1.7910070E 02	A64 = -9.8256370E 04
A65 = -2.8442070E 01	A66 = -9.9339400E 00	A67 = 1.4317490E 01
A68 = 5.9159390E-02	A69 = 2.1628560E 02	A73 = 2.5403440E-03
A77 = -8.6705500E-02	A78 = 2.5631110E-04	A81 = 3.2986130E 05
A82 = 8.3333390E 02	A84 = 4.4097750E 05	A85 = 1.1470870E 02
A86 = -3.6026590E 01	A88 = -6.4373300E-01	A89 = -9.6481830E 02
A94 = 4.6655270E-01	A95 = -2.5236600E-03	A96 = -2.9802320E-07
A98 = 6.5586110E-04	A99 = -2.2059300E-01	A104 = -1.2888150E-01
A105 = 6.9713330E-04	A106 = -7.2012910E-04	A108 = -8.4594270E-06
A109 = 2.8198090E-04		

B. B MATRIX.

B11 = -3.9051580E-02	B32 = 3.4094850E 00	B33 = 6.9783800E-02
B44 = -3.7504610E-04	B54 = -5.3618420E 01	B64 = -7.7428660E 01
B72 = 7.3857530E 00	B73 = 1.5741700E 00	B84 = 3.6026590E 01
B94 = -9.1892480E-02	B104 = 7.2012910E-04	

* Note: Only non-zero elements are listed.

C. C MATRIX.

C11 = 9.5000000E 04 C12 = 2.4000000E 02 C21 = -1.3194440E 02
C22 = -3.3333340E-01 C28 = 1.4152740E-03 C31 = 1.8049990E 00
C32 = 4.5549940E-03 C410 = 1.0000000E 00

D. D MATRIX.

D31 = 4.2000000E-01

E. STATE VECTOR

X_1 = superheater outlet density (lb/ft^3)
X_2 = superheater outlet temperature (°R)
X_3 = superheater tube-wall temperature (°R)
X_4 = quality of mixture leaving riser
X_5 = riser mass-flow rate (lb/sec)
X_6 = downcomer mass-flow rate (lb/sec)
X_7 = riser tube-wall temperature (°R)
X_8 = drum pressure (lb/ft^2)
X_9 = drum and downcomer liquid temperature (°R)
X_{10} = drum liquid level (ft)

F. INPUT VECTOR

U_1 = throttle opening (%)
U_2 = fuel mass-flow rate (lb/sec)
U_3 = air mass-flow rate (lb/sec)
U_4 = feedwater mass-flow rate (lb/sec)

G. OUTPUT VECTOR

Y_1 = superheater outlet pressure (lb/ft^2)
Y_2 = steam mass-flow rate from drum into superheater (lb/sec)
Y_3 = steam mass-flow rate at the superheater outlet (lb/sec)
Y_4 = drum liquid level (ft)

APPENDIX B

* REDUCED ORDER MODEL MATRICES ELEMENTS

A. A MATRIX.

A11 = -1.2436020E 01	A16 = 1.3159200E-04	A21 = -3.4339410E 00
A22 = -4.7662320E 00	A23 = -1.1941000E-03	A24 = -7.0739000E-04
A25 = 7.2198000E-04	A26 = 2.7080400E-06	A31 = -4.9093270E 05
A32 = -6.3842140E 05	A33 = -1.5757540E 02	A34 = 9.3258980E 02
A35 = 9.9146950E 01	A36 = 4.0967170E-01	A41 = -7.0894060E 04
A42 = -9.8256370E 04	A43 = -2.8442070E 01	A44 = -9.9339400E 00
A45 = 1.4317490E 01	A46 = 5.9159300E-02	A55 = -8.6705000E-02
A56 = 2.5631100E-04	A61 = 3.2986130E 05	A62 = 4.4097775E 05
A63 = 1.1470870E 02	A64 = -3.6026590E 01	A66 = -6.4373300E-01
A72 = -1.2888150E-01	A73 = 6.9713300E-04	A74 = -7.2012000E-04
A76 = -8.4594000E-07		

B. B MATRIX.

B22 = -3.7500000E-04	B32 = -5.3618420E 01	B42 = -7.7428660E 00
B51 = 3.3457900E 01	B62 = 3.6026590E 01	B72 = 7.2012900E-04

C. C MATRIX.

C16 = 1.0	C27 = 1.0	C31 = 1.0
C42 = 1.0	C53 = 1.0	C64 = 1.0
C75 = 1.0		

D. D MATRIX.

D81 = 1.0	D92 = 1.0

*Note: Only non-zero elements are listed.

E. STATE VECTOR

X_1 = superheater outlet density (lb/ft^3)
X_2 = quality of mixture leaving riser
X_3 = riser mass-flow rate (lb/sec)
X_4 = downcomer mass-flow rate (lb/sec)
X_5 = riser tube-wall temperature (°R)
X_6 = drum pressure (lb/ft^2)
X_7 = drum liquid level (ft)

F. INPUT VECTOR

U_1 = fuel-air mass-flow rate (lb/sec)
U_2 = feedwater mass-flow rate (lb/sec)

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